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Traction Launch of Scaled Laboratory Kinetic Energy Penetrators

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13. ABSTRACT *(Maximum 200 words)*

This report discusses the design, analysis, and testing phases of a modification in subscale, kinetic energy projectile technology. Typically, subscale penetrators are push launched, while the actual full-scale penetrators are traction launched. This condition was acceptable until recently when push launch technology could no longer fully satisfy the needs of the researcher. Therefore, a technique for implementing traction launch technology into the subscale laboratory environment was needed. This report presents the theory, finite element analysis, and actual testing of one concept. The design process was highlighted by the use of several finite elements codes (implicit and explicit) and the exceptional agreement between the results.

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1. INTRODUCTION

The most effective weapon of the modern tank against heavy armor threats is the kinetic energy (KE) projectile (Figure 1). This projectile consists of a subcaliber penetrator which is carried through the cannon by a multipetal sabot. The high inertial loads of the penetrator are transmitted to the sabot via a geometric traction interface (i.e., annular buttress grooves and/or a threaded friction drive).

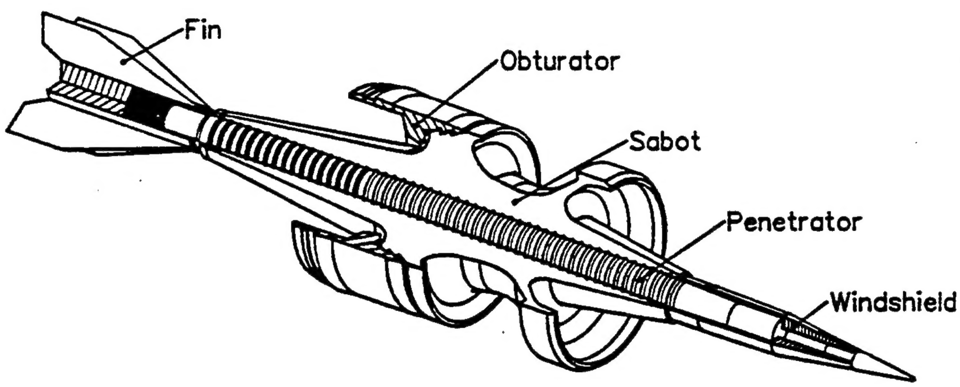


Figure 1. An isometric view of a fielded kinetic energy projectile.

However, in a research environment where one-quarter and one-third scale work is often performed, these shear transferring mechanisms present several distinct disadvantages. Complex penetrator geometry can require additional complexity in the sabot with increased manufacturing costs for the projectile. Due to integration and manufacturing considerations, some of the groove characteristics cannot be faithfully scaled; the typical affect is an increase in the percentage of groove mass with respect to the entire penetrator mass. Furthermore, since subscale tests are typically used to examine phenomenological trends, the presence of grooves can greatly increase the complexity of an analysis and can mask important factors.

Reducing the previous arguments to cost and simplicity, subscale penetrators have traditionally been right circular cylinders (except for different nose shapes) and are push launched. However, if the combination of penetrator geometry and material properties prohibit a push launch, a traction sabot becomes necessary. (Figure 2 illustrates the conceptual differences between push and traction launch

technology.) Therefore, a study was performed to design a sabot and a penetrator-sabot interface which would minimize the interface geometry for these subscale projectiles.

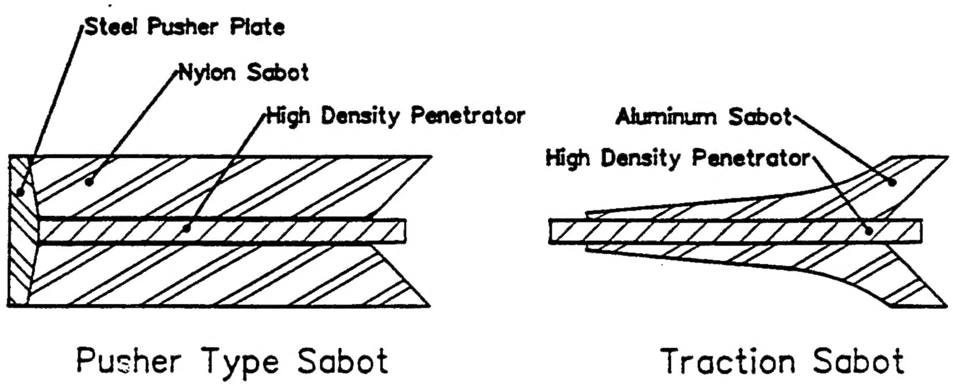


Figure 2. Comparison between push-launch and traction-launch sabot technology.

2. THEORY

The simplest form of traction is friction. However, it is not intuitively obvious that frictional forces alone can hold the penetrator in the sabot when the assembly is experiencing 50,000–150,000 g's of acceleration. Yet, the simplicity of the solution is attractive enough to merit further investigation.

As with any new design effort, simplifying assumptions are made and free body diagrams are generated to provide insight and governing equations for the problem. The geometry assumptions used in this analysis are that the penetrator is a cylinder and the sabot is a frustum (Figure 3). It is also assumed that the sabot fills the bore of the cannon and a pressure P is accelerating the projectile at $\ddot{z}$. The free body diagrams for each body appear in Figure 4.

Since the projectile is axisymmetric, radial forces cancel out, leaving force balances on the axial forces only. Using Newton's Second Law, force balances on the penetrator and sabot yield Equations 1 and 2, respectively. For the frictional concept to work, the acceleration of each body must be equal, and the same as the acceleration of the system. Performing a force balance on the system, the internal forces cancel, and Equation 3 is obtained.

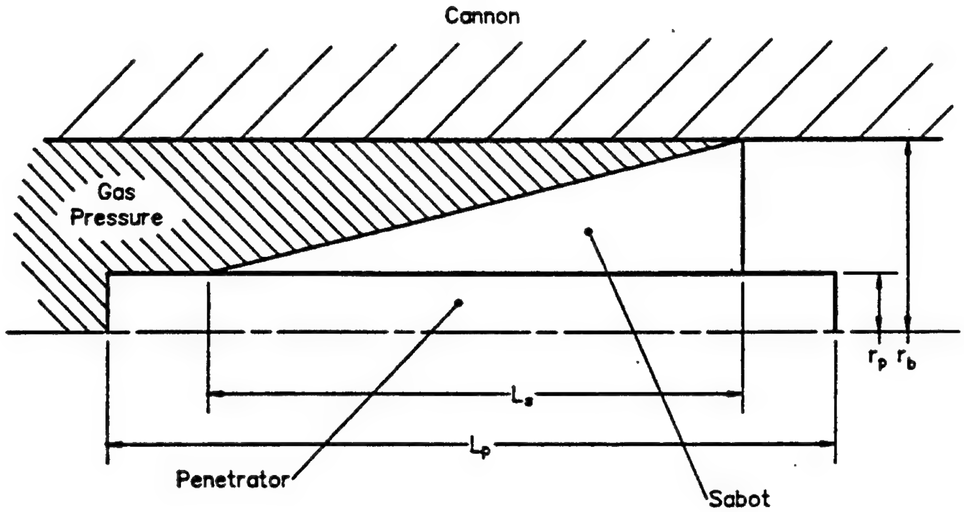


Figure 3. Simplified projectile model.

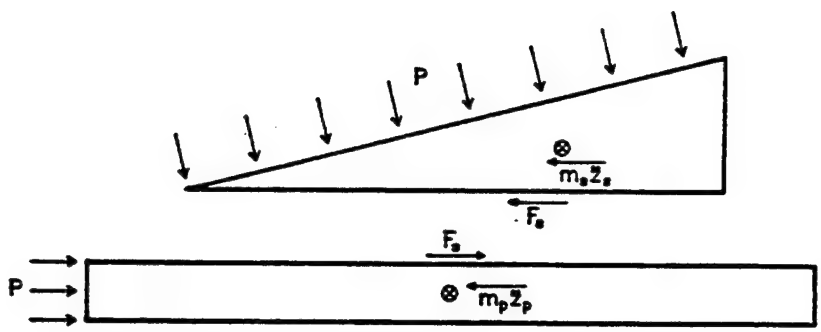


Figure 4. Free body diagrams.

$$m_p\,\ddot{z}_p = \pi\,r_p^2\,P + F_s \tag{1}$$

$$m_s\,\ddot{z}_s = \pi\,(r_b^2 - r_p^2)\,P - F_s \tag{2}$$

$$\ddot{z}_s = \ddot{z}_p = \ddot{z} = \frac{\pi\,r_b^2\,P}{m_s + m_p} \tag{3}$$

where,

m_p ≡ penetrator mass

m_s ≡ sabot mass

$\ddot{z}_p$ ≡ acceleration of penetrator c.g.

$\ddot{z}_s$ ≡ acceleration of sabot c.g.

r_p ≡ penetrator radius

r_b ≡ barrel radius

P ≡ pressure

F_s ≡ interface shear force

By subtracting Equation 2 from Equation 1 and removing the acceleration term by substitution of Equation 3, the governing equation can be written in terms of component masses, m_s and m_p, penetrator radius, r_p, barrel radius, r_b, pressure, P, and the term specifying the shear force between the bodies, F_s. It is the expression for the shear forces which will dictate the feasibility of the concept. To reduce Equation 4 into its fundamental parameters, Equations 5 and 6 provide the mass relationships for both the penetrator and the sabot. Equation 7 provides the physical foundation for F_s.

$$(m_p - m_s)\,\frac{\pi\, r_b^2\, P}{m_s + m_p} = \pi\, P\,(2\, r_p^2 - r_b^2) + 2\, F_s \tag{4}$$

$$m_p = \pi\, r_p^2\, L_p\, \rho_p \tag{5}$$

$$m_s = \frac{\pi}{3}\, L_s\,(r_b^2 + r_p\, r_b - 2\, r_p^2)\, \rho_s \tag{6}$$

$$F_s = 2\, \pi\, r_p\, L_s \bar{\sigma}_r\, \mu \tag{7}$$

where,

ρ_p ≡ penetrator density

ρ_s ≡ sabot density

L_p ≡ penetrator length

L_s ≡ sabot length

$\bar{\sigma}_r$ ≡ average radial stress at the interface

μ ≡ static coefficient of friction

less than 1.0 produces a feasible design since the required sabot length is less than the penetrator length. As mentioned earlier, the designer has some control over *f*, so it was also varied to provide a family of curves.

The results of this analysis are provided in Figure 5 for penetrator aspect ratios of 20, 30, and 40 for a 50-mm cannon. Materials used in the analysis were tungsten heavy alloy (WHA) for the penetrator and aluminum for the sabot, with densities of 17,600 and 2,700 kg/m^3, respectively. These results show that for $f \geq 0.3$, the concept is feasible if the assumptions are valid.

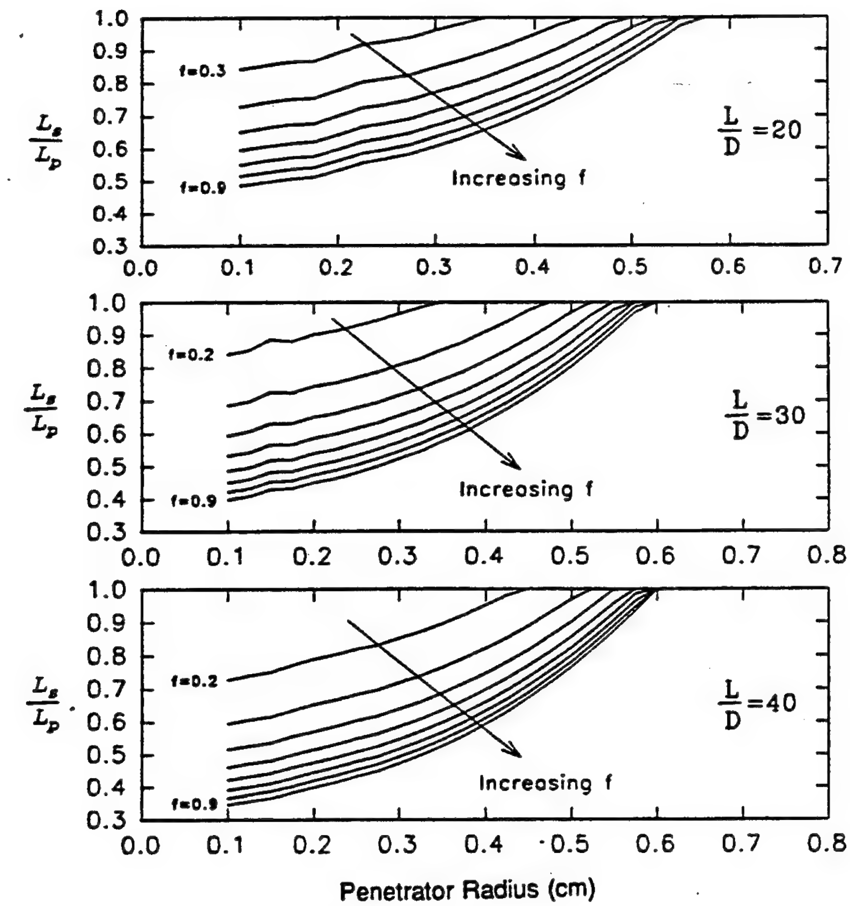


Figure 5. Results from Equation 8 representing minimum sabot length, normalized by penetrator length, to successfully launch a penetrator of a given radius from a 50-mm cannon.

In Equation 7, the term $2\pi r_p L_s$ represents the area of the interface between the sabot and the penetrator, μ is the coefficient of friction, and $\bar{\sigma}_r$ is the average radial stress at the interface. Assuming that the penetrator geometry is defined, F_s is a function of sabot length, L_s, interface stress, $\bar{\sigma}_r$, and the coefficient of friction, μ. $\bar{\sigma}_r$ is a function of sabot shape, material selections, and pressure, all of which the designer has some degree of control over. μ is a physical constant which is a function of the materials and surface quality. Therefore, the only remaining unknown is L_s. Substituting Equations 5–7 into Equation 4 and solving for L_s provides:

$$A\,L_s^2 + B\,L_s + C = 0 \tag{8}$$

where,

$$A = \frac{4\,\pi}{3}\,\rho_s\,(r_b^2 + r_p\,r_b - 2\,r_p^2)\,r_p\,\frac{\bar{\sigma}_r}{P}\,\mu \tag{8a}$$

$$B = \frac{2\,\pi}{3}\,\rho_s\,(r_b^2 + r_p\,r_b - 2\,r_p^2)\,r_p^2 + 4\,\pi\,r_p^3\,L_p\,\rho_p\,\frac{\bar{\sigma}_r}{P}\,\mu \tag{8b}$$

$$C = 2\,\pi\,r_p^2\,L_p\,\rho_p\,(r_p^2 - r_b^2) \tag{8c}$$

The values for r_p, L_p, ρ_p, ρ_s, and r_b are constants. The terms $\bar{\sigma}_r$, P, and μ are parameters which the designer can use to control L_s. For analysis purposes, these three terms are lumped together as term f.

$$f = \frac{\bar{\sigma}_r}{P}\,\mu \tag{9}$$

To determine the feasibility of the concept, the following parametric study was proposed. The variables ρ_p, ρ_s, and r_b were defined for a system of interest. To study an entire class of penetrators, aspect ratio (L/D) was defined. Equation 8 was then solved by holding f constant and varying r_p over the range of interest. To provided a measure of goodness to L_s, it was normalized by L_p; thus, any result

3. ANALYSIS

The remaining portion of this report will concentrate on verification of the theory through finite element and experimental methods. To limit the scope of this initial finite element study, only one cannon bore diameter will be examined—50 mm.

Because of the nature of the curves in Figure 5, the parametric analysis capabilities in ANSYS (DeSalvo and Gorman) were especially useful in examining the assumptions made earlier. A parametric model of Figure 3 was constructed in ANSYS, and more than 100 different projectile geometries were examined by varying L_p and r_p and calculating L_s. For the purpose of this study, f was assumed to be 0.3. The finite element model was axisymmetric and the solution was performed quasi-statically (Drysdale 1981). Nodes along the penetrator-sabot interface were shared to form a simple bimetallic interface. Proper boundary conditions, pressure and acceleration, were computed and applied for each set of geometry. In addition, one node on the penetrator was fixed in the axial direction, as required for an axisymmetric, static solution.

Using the results from these analyses, two points can be made. First, the expression $f = 0.3$ can be checked for validity. The coefficient of friction is a physical constant; therefore, only the expression $\bar{\sigma}_r/P$ needs to be evaluated. For each configuration, $\bar{\sigma}_r$ along the material interface was computed. By choosing a reasonable value for μ, f can now be defined. Results of the parametric analysis put $\bar{\sigma}_r$ at 120–150% of P (Table 1). From handbook tables (Oberg, Jones, and Horton 1985), μ is at least 0.3–0.4; therefore, a value of f ~0.3–0.5 is reasonable to expect and, according to Figure 5, many penetrator configurations exist for the specifications originally set forth.

In addition to examining the relationship between $\bar{\sigma}_r$ and P along the sabot/penetrator interface, another relationship exists between the radial and shear forces. Even though the nodes on the material interface are shared, significant information can be extracted. By comparing the product of the radial stress and coefficient of friction, $\sigma_r\mu$, to the shear stress, τ, the status of a sliding interface can be inferred. If the product of the coefficient of friction and the radial stress is larger than the shear stress, sufficient traction exists to accelerate the penetrator and sabot together. To provide a measure between the two parameters, each was plotted along the length of the sabot (Figure 6) for a group of penetrators (L/D = 30). Figure 6 illustrates that adequate traction exists to accelerate both bodies together.

Table 1. Result of $\bar{\sigma}_r/P$

R (m)	L/D				
	20	25	30	35	40
0.0020	1.64	1.54	1.48	1.42	1.40
0.0025	1.47	1.41	1.37	1.34	1.31
0.0030	1.38	1.34	1.31	1.28	1.26
0.0035	1.32	1.29	1.27	1.25	1.23
0.0040	1.29	1.26	1.24	1.22	1.22
0.0045	1.26	1.24	1.22	1.21	1.21
0.0050	1.24	1.22	1.21	1.20	1.20
0.0055	1.22	1.21	1.20	1.20	1.21
0.0060	1.21	1.20	1.19	1.20	1.20
0.0065	1.20	1.19	1.18	1.19	1.19
0.0070	1.19	1.18	1.19	1.19	1.19

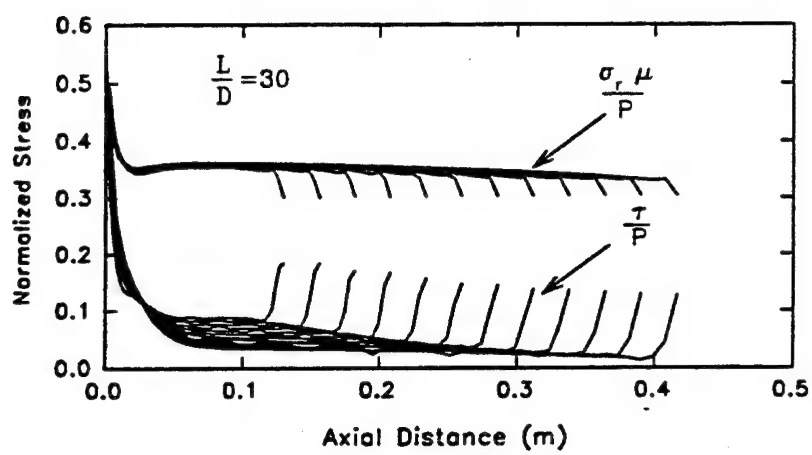


Figure 6. Comparison of normalized radial and shear stress along the penetrator-sabot interface for L/D 30 penetrators.

Although the results of the previous analyses tend to support the initial premise that frictional forces alone will provide sufficient traction to the penetrator, two significant assumptions were made: the penetrator-sabot interface used shared nodes and the analyses were quasi-static. To continue toward a final

design and manufacture projectiles would be imprudent until these assumptions were removed. Therefore, a dynamic analysis with some model of the sliding interface was necessary.

To conduct a dynamic analysis in ANSYS was very difficult with the present capabilities. ANSYS Revision 4.4A has the necessary capabilities to model the sliding interface by using the 2-D interface element (STIF12). However, the introduction of friction adds nonconservative forces into the analysis and the rate at which loading occurs (from 0 to 400 MPa in 2 ms) forces an extremely small time step to ensure convergence. Providing a mesh which would accurately resolve the stresses would severely tax the workstation which ANSYS is resident to the point where one analysis would take days or even weeks to complete. Therefore, the NIKE2D and DYNA2D hydrocodes were utilized in a supercomputer environment. NIKE2D and DYNA2D (Hallquist 1983, 1986, 1988) are two hydrocodes which were developed at the Lawrence Livermore National Laboratory to solve dynamic problems. The codes are similar with the exception of their integration algorithms; NIKE is implicit and DYNA is explicit. Both codes have been vectorized to execute efficiently in the CRAY environment and can also resolve sliding interfaces more efficiently than ANSYS (Rev 4.4A) can.

To demonstrate that the sabot concept works, prototype projectiles needed to be manufactured and tested. To work towards this objective, the remaining finite element modeling efforts focus on the candidate projectile. The penetrator is cylindrical with a length of 0.24 m and a diameter of 0.008 m. The sabot length was determined from Equation 8 using a value of 0.3 for f. This is a conservative value for f, but the increased margin was desirable due to the novelty of the work.

The finite element mesh used in the NIKE and DYNA analyses is displayed in Figure 7. Both programs were used to provide confidence that different solution techniques yielded similar results. In both models, slidelines with voids and friction ($\mu = 0.3$) were utilized. Figure 8 displays the resulting displacement, velocity, and acceleration curves for the penetrator from the NIKE analysis. Although these curves provide information to compare to the DYNA analysis, the most telling information is penetrator displacement relative to the sabot (Figure 9). These curves show the relative axial displacement between the penetrator and the sabot at the three locations indicated. The results show that even though there is relative displacement, it is small enough to be inconsequential. To be complete, the results of the DYNA analysis are also presented (Figure 10). The agreement between the two solutions is quite good. For the location at the front of the sabot, there is a 7% difference in the maximum relative displacement and a

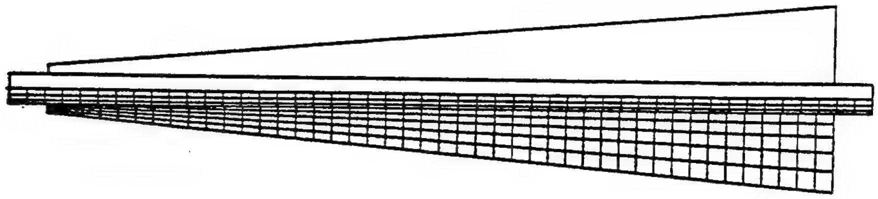


Figure 7. Mesh used in the NIKE and DYNA analyses.

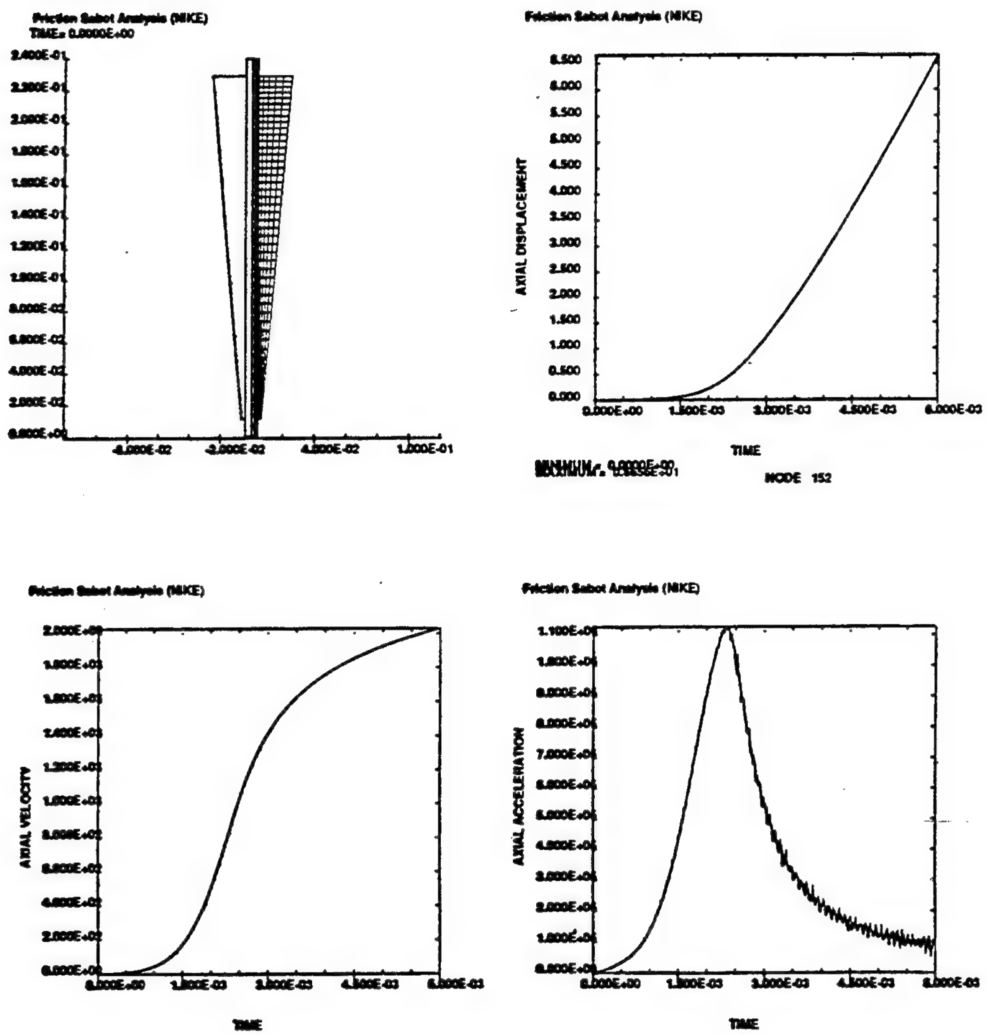


Figure 8. Displacement, velocity, and acceleration histories from NIKE.

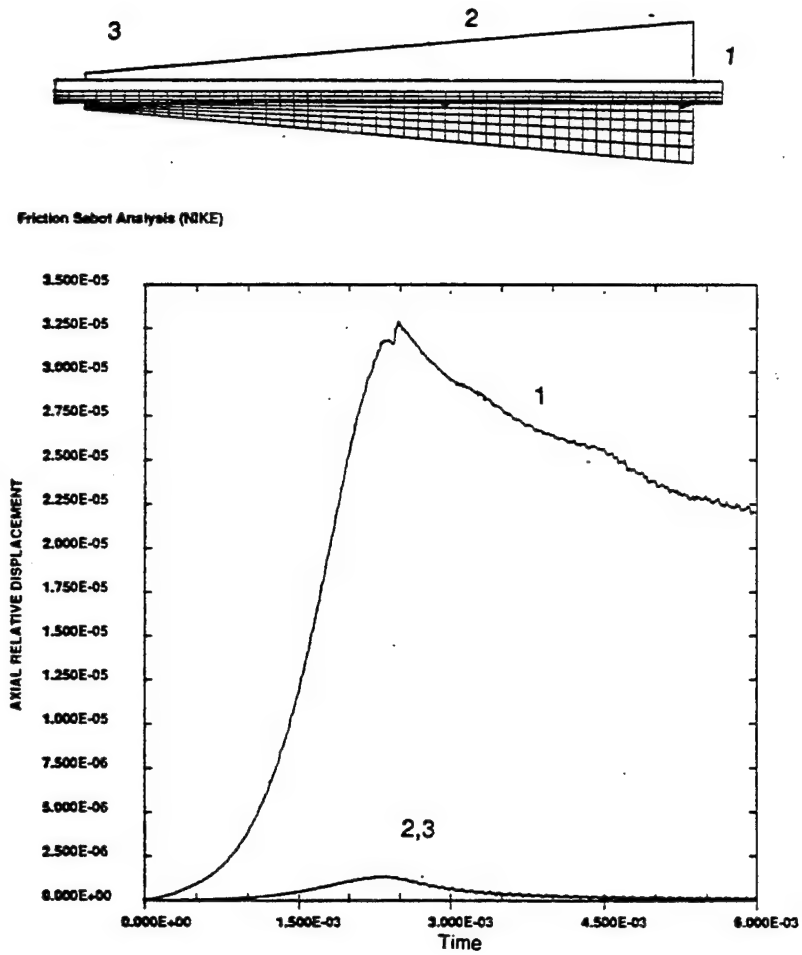


Figure 9. Relative displacement between nodes on the penetrator and the sabot at three different locations. (NIKE analysis, ITS = 5.0E-6.)

slight difference in the shape of the curve beyond the maximum. This is acceptable when considering that the integration time step in DYNA was an order of magnitude smaller.

The difference between curve 1 and curves 2 and 3 in Figure 9 and 10 is explained in Figure 11. This figure shows two curves, the product of the normal interface force and the coefficient of friction ($F_n \mu$) and the tangential interface force (F_s) as a function of axial distance from the front of the sabot. This graph shows that a small part of the interface does not have sufficient normal force to prevent sliding; therefore, the front of the sabot has a larger relative displacement. This was also the case with the quasi-static results from ANSYS shown in Figure 5. In the ANSYS analysis, examination of the front of the sabot shows that the shear stress is larger than the product of the radial stress and the coefficient of

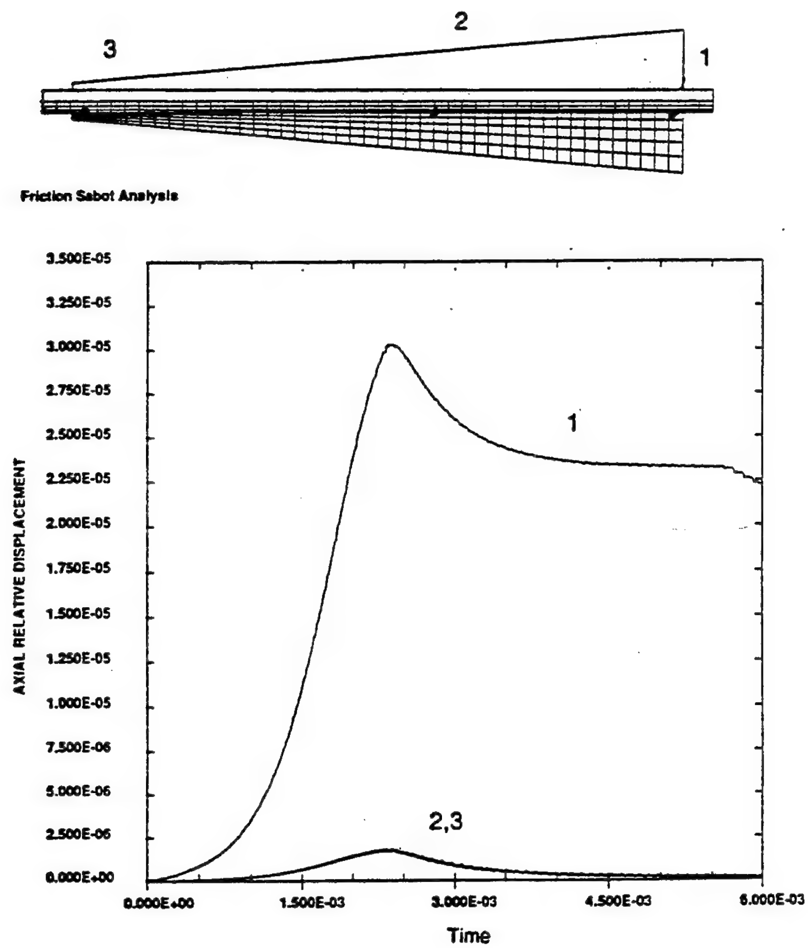


Figure 10. Relative displacement between nodes on the penetrator and the sabot at three different locations. (DYNA analysis, ITS = 5.0E-7.)

friction; therefore, if the nodes along the interface were not shared, slippage would occur. In the ANSYS analysis, the shear stress was higher and more localized because the nodes were shared and slippage could not occur as in the NIKE and DYNA analyses.

Based on the successful result of the dynamic analyses, a final sabot design was developed. This design incorporated all of the features necessary to interact with the cannon in an appropriate manner, be mass efficient, and not violate any of the initial assumptions. Two conditions which must be met within the cannon are stability and a seal to prevent gas leakage. A third condition must also be met at muzzle exit, a means to use aerodynamic forces to separate the sabot from the penetrator. Finally, the sabot must be inexpensive to manufacture. To maintain stability in the cannon, the sabot must have one contact point

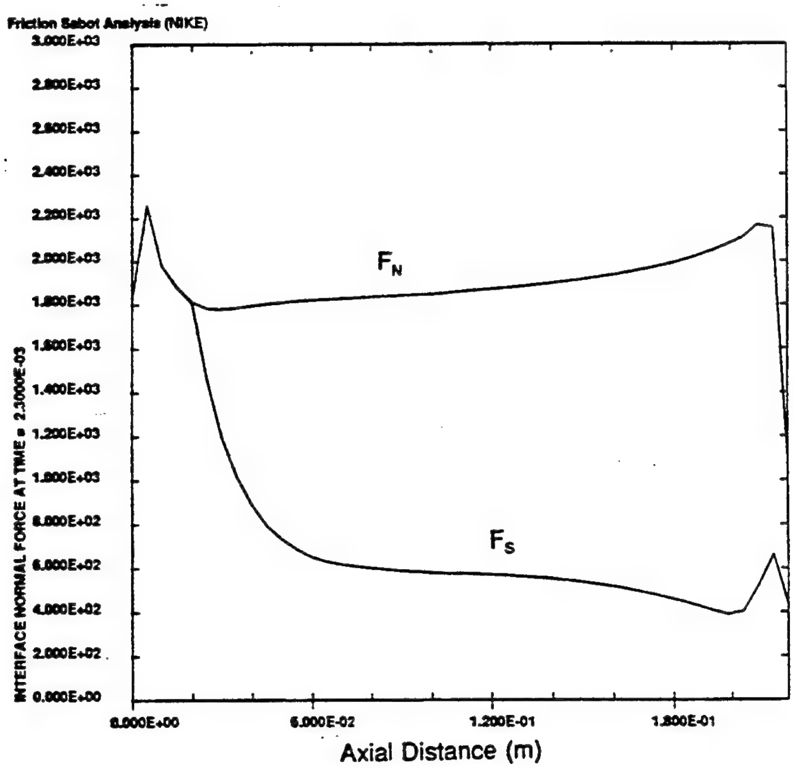


Figure 11. Sabot-penetrator interface forces at peak pressure.

at least two-thirds the bore diameter in length or two contact points separated by this distance. Because the sabot must be a single-ramp design (initial assumption), only one bore riding surface is possible. The required aerodynamic discard forces are generated by incorporating a scoop within the bore riding surface. Finally, a high-pressure seal is formed by machining notches into the bore riding surface and inserting nylon bands. By making the diameter of the nylon bands larger than the bore and pressing the projectile into the cannon, an effective seal is provided. Figure 12 shows a sketch of the proposed final design, illustrating the incorporation of the above details.

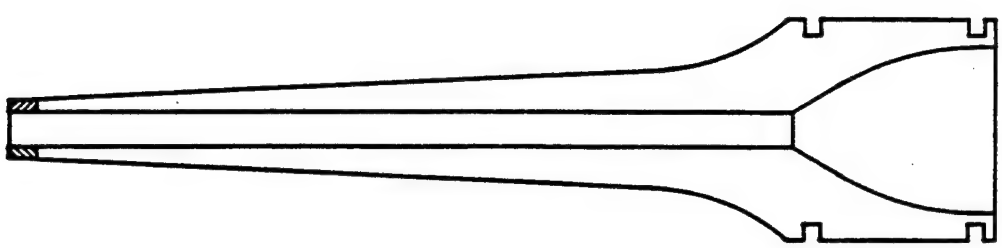


Figure 12. Final sabot concept.

Now that the sabot design has been conceptualized, final dimensions are required. To provide this information and minimize mass while ensuring that stresses did not exceed yield criteria, a parametric, quasi-static analysis was conducted in ANSYS. The groove details for the nylon bands were removed and a parametric model was generated for the sabot profile. The analysis was performed quasi-statically because dynamics were not a concern in determining the stress distribution within the projectile. Furthermore, since the sabot mass must be below a specified threshold and not the absolute minimum, the optimization algorithm was not utilized. The final stress results for the projectile for the quasi-static ANSYS analysis are presented in Figure 13. Also plotted are the NIKE results for the same projectile at peak pressure for a dynamic solution. The correlation between solutions is excellent, providing adequate verification of the results.

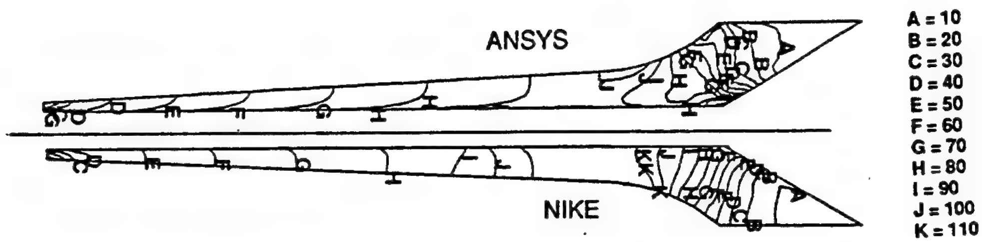


Figure 13. Effective stress contour plot of the final sabot design. Both ANSYS and NIKE results are presented in percentage of yield stress.

Two issues remain to be discussed—discarding the sabot at muzzle exit and manufacturing. To discard the sabot, it must be split into at least two parts along the axis of symmetry. This will permit the aerodynamic forces to separate the sabot from the penetrator. This poses several problems. First, if the sabot is in multiple pieces, there are more than two independent bodies which must be kept together while accelerating through the cannon. Second, manufacturing costs will increase. Finally, there is no static mechanism to hold the penetrator in the sabot prior to the launch sequence. To solve these problems, the following strategy was proposed. The sabot would be turned on a lathe in one piece. The inside details would be completed first, leaving a 0.02–0.04 mm interference with the penetrator at the aft end of the sabot. Once the turning operations were complete, wire EDM technology would be used to place one (or two orthogonal) splits in the sabot along the axis. These splits would start at the front of the sabot and extend to the aft, leaving 6–10 mm of length uncut. In this region, an interference fit with the penetrator would generate sufficient radial stress to hold the two components together while handling the projectile. To prevent gas from passing through the splits, a RTV (silicone rubber) seal was placed on the projectile in a manner to prevent any RTV from getting into the splits.

The only question remaining to be answered is how will the sabot break at discard? Will fracture occur along the axis of symmetry on each split, or will the fracture plane be perpendicular to the axis of symmetry? The desirable fracture plane is parallel to the axis of symmetry, but it is unknown whether this will happen. As a stress concentration exists due to the splits, the hypothesis is that fracture will continue in the direction of the splits. However, this will not be known until after initial testing has occurred.

4. TEST RESULTS

Two projectiles of two separate designs were tested for a total of four tests. The difference between the two designs was simply a manufacturing detail; all four projectiles had two orthogonal splits, but two were created by wire EDM and two by a band saw. The only substantial difference being the kerf for the band saw was much larger. A photograph of a projectile is presented in Figure 14. The front of the projectile is on the right and one of the splits is clearly evident. The two annular bands on the forward portion of the sabot are the nylon obturation bands. The penetrator is flush with the front of the sabot and extends past the aft end of the sabot.

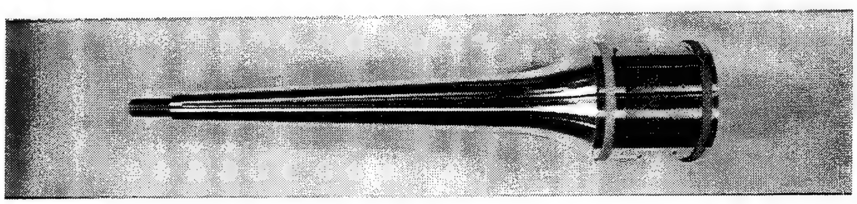
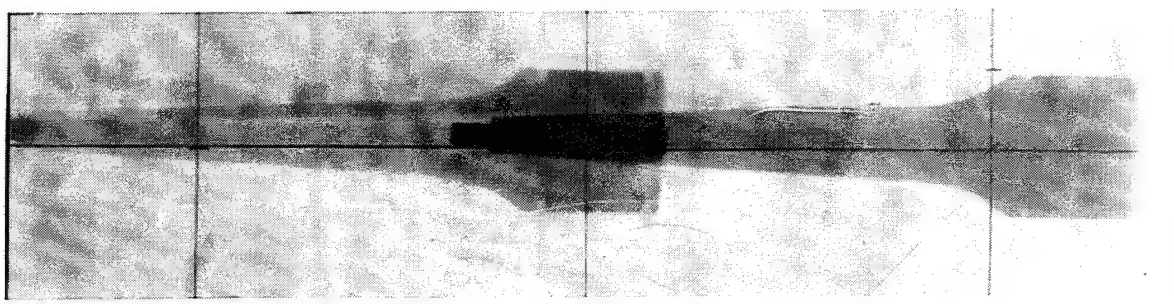


Figure 14. Photograph of the projectile.

The launch conditions for each test were kept similar. The maximum breech pressure was approximately 400 MPa, resulting in a maximum acceleration of 100,000 g's and a muzzle velocity of 1,650 m/s. A later test entry will increase the pressure to the cannon's upper limit of 650 MPa, resulting in a muzzle velocity in excess of 2,000 m/s. Of four tests, three were successful. The unsuccessful test was the first projectile with the band saw splits. In this test, the penetrator experienced setback and separated from the sabot while in-bore. One hypothesis for this traction failure the splits in the sabot were not deburred well enough, minimizing the contact area between the penetrator and sabot.

The confirmation of the concept working is provided by flash radiography. Figure 15a shows a radiograph of a static projectile. This figure shows the projectile configuration prior to loading it into the

cannon. Figure 15b shows two images of the projectile within 1.0 meter of muzzle exit. The direction of flight is left to right; several observations can be made. First, the penetrator is slightly ahead of the sabot; therefore, traction due to frictional forces was sufficient to accelerate the penetrator along with the sabot. Second, the right image shows more rotation of the sabot sections and that the fracture plane was parallel to the axis of symmetry and that complete mechanical disengagement of the sabot from the penetrator has occurred. This radiograph proves conclusively that the concept and analyses were valid.



A. Static radiograph of a projectile prior to loading.

B. Radiograph of a projectile at muzzle exit (velocity = 1,650 m/s).

Figure 15. Radiographs of the projectile.

5. CONCLUSIONS

To support the current and future needs of researchers in the area of penetration mechanics, a new concept for the traction launch of subscale KE penetrators has been proposed. By making simplifying assumptions, utilizing free body diagrams and simple physical relationships, a set of governing equations

were developed to specify sabot geometry based on penetrator and cannon geometry. Extensive use of finite element techniques were employed to verify the initial assumptions and concept feasibility.

During the finite element analysis, several different codes were used. In two different analyses, the parametric capabilities of ANSYS were utilized to great advantage in quasi-static solutions. However, ANSYS did not provide adequate means to solve the dynamic solutions. For these problems, NIKE and DYNA were utilized and provided sufficient confidence to continue to the final design. Lastly, comparison of results from quasi-static (ANSYS) to dynamic (DYNA) solutions was exceptional.

Utilization of finite element methods verified the theory presented earlier. However, verification of both the theory and finite element analysis can only be accomplished by actual tests. The outcome of the tests proved that the concept, assumptions, and analysis process were valid.

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